Ministry of Higher Education
and Scientific Research
University of Diyala
College of Engineering

EXPERIMENTAL AND NUMERICAL INVESTIGATION FOR THERMAL ENHANCEMENT OF SHELL AND COIL HEAT EXCHANGER

A Thesis Submitted to the Council of College of Engineering,
University of Diyala in Partial Fulfillment of the
Requirements for the Degree of Master of Science in
Mechanical Engineering

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2021 A.D IRAQ 1442 A.H

بسم الله الرحمن الرحيم

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I certify that this thesis entitled **(Experimental and Numerical Investigation for Thermal Enhancement of Shell and Coil Heat Exchanger)** had been carried out under our supervision at the University of Diyala / College of Engineering - Mechanical Engineering Department in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering.

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ACKNOWLEDGEMENTS

All praises to Allah for the strengths and His blessing in completing this thesis.

I would like to express my appreciation and gratitude to my supervisors, ***Prof. Dr. Anees A. Khadom*** *and* ***Asst. Prof. Dr. Itimad D. J. Azzawi*** *for their supervision and constant support, their invaluable help of constructive comments and suggestions throughout.*

I express my special thanks ***Asst. Prof. Mustafa S. Mahdi*** *from Chemical Engineering Department for his efforts and support throughout my research.*

Also, I am extremely grateful to my parents, brothers and sisters for their love and sacrifices for educating and preparing me for my future.

Finally, my thanks go to all the people who have supported me to complete the research work directly or indirectly.

Senaa Khazal Ali

2021

ABSTRACT

The design of shell and helical coil heat exchangers is very important subject in industrial processes due to their simple design, low cost production and highly efficient. Many researchers studied the enhancing of heat transfer by using the helical coils that widely used in industrial applications, such as food and chemical process, power generation, electronics and nuclear industries, etc. Additionally, it can generate the secondary flows by centrifugal forces without having any moving part, which leads to enhancement of heat transfer rate.

This work is an experimental and numerical studies to evaluate how geometrical parameters such as coil diameter (D_c), tube diameter (d_i) and pitch coil (p) influencing heat transfer of shells and coils heat exchangers. The experiment initially performed to verify the results of numerical analysis on shell and single coil (baseline coil) heat exchanger, numerical analysis applied to determine the effect of changing the baseline coil to new double coil configuration on heat transfer enhancement. A numerical study was conducted to analyze the flow structure inside the single and double coil by using the RNG k-ε turbulence model at various coil pitch. Finally, multi objective optimal design of the coil pipe was performed.

The results showed a good agreement with experimental results with an error equal to numerical 4.6%. In addition, the numerical simulation results were conducted on a double coil and found that the Nusselt number in a double coil tube was 18.2% greater than that in a single coil tube at 1800 Reynold number of the shell side. Based on the results, the two correlations were developed to predict the Nusselt numbers of shell and coil sides for wide ranges of Reynolds number ($400 < Re_{sh} < 2000$) ($11000 < Re_c < 22000$). Meanwhile, a friction factor inside helical coil increases of 16.5% is obtained in the double coil than single coil. A correlation of the Reynolds number coil side as a function of friction factor inside the helical coil was presented.

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NOMENCLATURE

Symbol	Definition	Unit
A	area	m^2
Cp	specific heat	J/kg. ℃
d	tube diameter	m
D_c	curvature diameter	m
D_h	equivalent diameter (hydraulic diameter)	m
H	coil height	m
h	convection heat transfer coefficient	W/ m^2
k	thermal conductivity	W/m. ℃
L	tube length	m
$m^{\cdot}$	mass flow rate	kg/s
N	number of turns of the helically coiled tube	
Nu	Nusselt number, dimensionless	
P	coil pitch	m
Q	heat transfer rate	W
T	water temperature,	℃
t	tube wall thickness	m
U	overall heat transfer coefficient	W/m^2. ℃
u	velocity	
$V^{\cdot}$	volumetric flow rate	m^3/s
ΔP	pressure drop	Pa
	Subscripts	
c	coil side	
co	cold	
h	hot	
i	inlet/inner	
o	outlet/outer	
sh	shell side	
	Greek symbols	
γ	dimensionless pitch ratio (coil torsion), $(\frac{p}{\pi D_c})$	
δ	dimensionless, ratio outer or inner diameter of first and second coil	
ρ	density	kg/m^3
μ	dynamic viscosity	kg/m. s
ε	effectiveness	
τ_{ij}	Viscous stress tensor	
S	The modulus of rate-of-strain tensor	
α_k	The Prandtl number for turbulent kinetic energy.	
α_ε	The Prandtl number for turbulent dissipation rate.	
G_k	generation of turbulence kinetic energy due to the mean velocity gradients	
	Abbreviation	
LMTD	Log mean temperature difference	
NTU	Number of transfer unit	
CFD	Commotional fluid dynamic	

SIMPLE	Semi Implicit Method	
FVM	Finite Volume Method	
SWHE	Spiral wand heat exchanger	
	Non dimensional groups	
f	friction factor, dimensionless	
Nu	Nusselt number, dimensionless	
Pr	Prandtl number, dimensionless	
Re	Reynolds number, dimensionless	
De	Dean number, dimensionless	

Chapter One

Introduction

1.1 General Introduction

Due to the significant and persistent growth in energy consumption rates and the rising scarcity of conventional energy supplies and high prices, the energy crisis is considered one of the most critical problems facing the world. Researchers are thus deliberately trying to improve the performance of heat exchange systems and reduce their size in order to reduce their rates of energy usage.

Resources and energy economic considerations have provided an opportunity to increase efforts to manufacture more efficient heat exchangers in term of thermal energy consumed. Moreover, some applications used in space and aviation are essential factors that should be concerned with examining the scale and weight of the heat exchanger.

The heat exchanger is a system used to transfer thermal energy between two fluids, where the fluids can be separated by a solid wall to prevent mixing (indirect contact) or can be in direct contact with each other [1]. A variety of types of heat exchangers are available. Every form has its own advantages and disadvantages. Active and passive methods are widely used to improve the coefficient of heat transfer [2]. In a heat exchanger, heat transfer generally includes convection in each fluid and conduction through a wall which separates the two fluids.

1.2 Classification of Enhancement Techniques

Numerous techniques have been used to improve heat transfer and these techniques can be divided into three main groups [3-5]:

1. passive techniques: this technique does not require an external power and is commonly used with additional devices or inserts by geometric or surface changes in the flow channel with.
2. Active techniques: this technique is more complex than passive design techniques and application, because external energy is needed to regulate the flow of fluids to improve thermal efficiency.

The use of active science techniques is limited, as it is not easy to supply an external energy in most applications.

3. Finally, the conjunction between passive and active techniques such as rough surface with fluid vibration and which are called compound techniques [6].

When designing any heat exchanger, the heat transfer optimization technique can be used to optimize the features of the heat exchanger for different purposes. Where such changes include reducing the size, lowering the surface temperature and increasing the reliability and workability of the process. Nonetheless, the choice of the optimum method for optimizing heat transfer is controlled by a number of conditions that should be taken into account in each specific case [7].

1.3 Helically coiled tube and design

In an effort to get same amount of heat transfer as a straight tube heat exchanger in a small room, the straight inner pipe replaced with a helical coil. That allows more heat transfer in a smaller shell surface area, while the pressure drop through the heat exchanger increases. Owing to the geometrical configuration of helical coils, the helical coil has a more complicated flow pattern. They are one of the most common forms of passive technology that produces comparatively more compact heat exchangers used in a wide variety of heat transfer applications [8]. This is due to high efficiency and compact size compared with straight tubes. The flow field and the overall coefficient of heat transfer in the helically coiled tube are complex compared to the conventional heat exchanger and this is due to the dependence of the secondary flow behavior on the curvature of the tubes. Such secondary flows are responsible for high turbulence, which enhances the rate of heat transfer in coiled tubes [9]. Besides, the centrifugal force is generated within the fluid flow due to the curvature of the coil tube so that, the heat transfer rate is significantly increased as the secondary flow induced.

The helically coil has an inner diameter (d_i) with a curvature diameter of coil (D_c) (centerline diameter), while the distance between two adjacent turns, called pitch (p) as shown in Figure 1.1. The curvature ratio (d_i / D_c) is the internal diameter to coil diameter ratio, which is responsible for producing a secondary flow that induces centrifugal force when the fluid flows through the coil as shown in Figure 1.2. Thus, the variation in heat transfer and flow characteristics between the helical coil and the straight tube is primarily associated with the secondary flow effect [10]. This secondary flow has a significant ability to improve the heat transfer rate by making fluid in the middle of the curved tube moving centrifugally outwards, and fluid near the wall inwards.

The curved duct problem was first proposed mathematically by Dean (1927) under clearly described flow conditions. For the fully developed flow within a curved circular tube, the presence of a pair of counter-rotating vortices as a secondary flow in the curved tube was confirmed [11]. In order to verify the distribution of velocity, pressure field and secondary flow at various coil parameters, Lingdi el at [12] investigates the numerical efficiency of flow characteristics inside the helical coil (i.e., Dean number, curvature radius, and coil pitch). Results revealed that the velocity gradient increases in the helical coil tube and the secondary flow is the main reason for reduction of flow causing turbulent flow in the coil. Therefore, the Dean number (De) can be defined as a dimensionless number giving the ratio of the viscous force acting on a fluid flowing in a curved pipe to the centrifugal force, equal to the Reynolds number (Re) times the square root of the ratio of the diameter of the coil tube (d_i) to its curvature diameter (D_c) [13]:

$$De = Re\left(\sqrt{\frac{d_i}{D_c}}\right) \tag{1.1}$$

$$Re = \frac{\rho v d}{\mu} \tag{1.2}$$

Where ρ is the density of liquid, v is the velocity along the tube, d is the diameter (hydraulic diameter (D_h) for shell side or inner diameter for coil tube side) [14], and μ is the dynamic viscosity.

$$D_h = \frac{D^2 - \pi D_c d_0^2 \gamma^{-1}}{D + \pi D_c d_o \gamma^{-1}} \tag{1.3}$$

According to the analysis by Ghorbani et al [15], the critical Reynolds number of the helical coil fluid that chooses to flow is laminar or turbulent is related to the curvature ratio as follows:

$$Re_{crit} = 2100\left[1 + 12\left(\frac{di}{D_c}\right)^{0.5}\right] \tag{1.4}$$

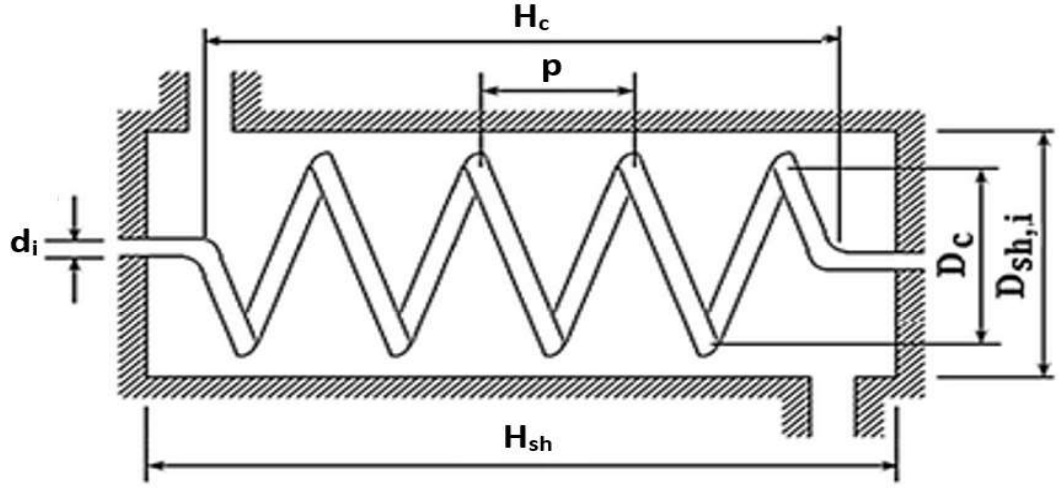

Fig. 1.1. Schematic of Heat Exchanger Shell and Helical Coil Tube [16].

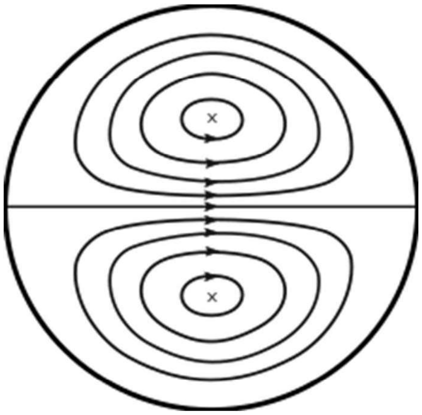

Fig. 1.2. Region of secondary flow in helical coiled tubes [17].

Since the coil tube is compact, it requires a smaller space. The choice usually falls out on the coiled tube heat exchangers in order to achieve the maximum thermal exchange. It is also simple to produce and can be operated at high pressure and also is suitable for use on shell side under laminar flow conditions or low flow rates. Thus, these heat exchangers can sometimes be very cost-effective [18]. The focus has thus begun to be of further improving the thermal performance of helical coiled tube heat exchangers which is dependent on numerical analysis using various computing software (Computational fluid dynamics (CFD)).

Computational power enhancements also increased the attention of engineers and scientists in simulating their problems using computational and numerical methods. In recent decades, several computing instruments and techniques have been developed to study fluid dynamics, combustion, and various heat transfer modes. Where many engineers are used Computational fluid dynamics (CFD) such as ANSYS CFX, ANSYS FLUENT or STAR-CD to solve the entire system in small cells, with apply governing equations (continuity, momentum and energy) to find numerical solutions with respect to temperature gradients, pressure distribution and velocity vector [19].

Computational fluid dynamics can be defined as a simulation method that uses powerful computers and applied mathematics to model fluid flow situations for heat transfer, mass and momentum prediction, and optimal design in industrial processes [20]. The most important advantage of the CFD approach is the possibility to model three dimensional geometries where it allows geometric improvements to be measured with much less time and expense than would be required in laboratory research [21]. In addition, it gives a deep understanding of the distribution of flow, the movement of mass and heat, the separation of particles, etc. As a consequence, both of these will provide a much clearer and broader picture of what happens in a given phase [20].

In the recent past few years, a wide range of experimental and numerical literature have tried to improve the heat transfer rate by using single and double helical coil heat exchanger. CFD has emerged as a cost effective alternative and it provides speedy solution of any kind of analysis requirement ranging from prediction of fluid flow behavior to complete heat exchanger design and optimization involving a wide range of turbulence models and integrating schemes available in CFD softwares [22]. Itimad et al [23-24] examine the influence of laminar flows on the coil friction factor and wall shear stress numerically when the coil pitch varies from 0.01 to 0.025m. They observed good results at the coil pitch of 0.01m, noted that the 0.05m coil pitch was not significantly different from the 0.01m coil pitch, where the maximal pressure fell from 71 Pa for 0.01m coil pitch to 68.8 Pa for 0.05m coil pitch. Due to the extra helix length, the percentage of rise in pressure as the coil pitch increased from 0.25m to 0.05m is almost 47%. In order to demonstrate the influence of varying coil pitch on the coil friction factor and wall shear tension. Itimad et al [25] performed a theoretical investigation of turbulent flow within the helical coil. Two turbulence models used to test the turbulence model that could capture much of the flow characteristics were STD (k-ε) and STD (k-ω). The findings revealed that the Dean number had a greater effect on decreasing the coil's friction factor in turbulent flows than on increasing the pitch dimension.

1.4 Objectives and Overview

Most researchers concentrated on using single and number of coils inside one shell. The experimental and numerical investigations of the present work can be introduced with the following organizations:

- Firstly, build the experimental set up heat exchanger composed from shell and helical coil, and equipped with the necessary measurement instruments for temperature, and flow rates.
- Provide a validation study between the numerical and experimental results in terms of Nusselt number of shell side using shell with single coil (baseline case) heat exchanger.
- Build a model of the shell and double coil heat exchanger and compare two types of coil configurations, single and double on following important parameters at steady state conditions:
    - Nusselt number of shell side (Nu_{sh})
    - Nusselt number of coil side (Nu_c)
    - Friction factor inside helical coil (*f*)
- Study and investigate the flow structure of double coil for geometrical parameters (i.e. Curvature diameter, coil pitch and coil diameters) with a wide range of Reynolds number for shell and coil side ($400 < Re_{sh} < 2000$) and ($11000 < Re_c < 22000$).
- Establish the optimum configuration of double coil in terms of desirable heat transfer rate, multi objective optimal design of helically coil pipe is conducted as a function of second coil curvature diameter (D_{c2}), pitch coil (P) and diameter of the coil tube ratio ($\delta = \frac{d_{i,c1}}{d_{i,c2}}$).

- Study the flow structure inside single and double coil with a wide range of Reynolds number coil side ($11000 < Re_c < 22000$) and coil pitch (30, 60 and 90mm).
- Finally, suggest a correlation between the predicted and numerical Nusselt number of the shell side (based on hydraulic diameter) and coil side for wide ranges of Reynolds numbers in addition to the friction factor inside the helical coil.

Chapter Two

Literature Review

2.1 Background

In most industries, heat exchanger design and thermal evaluation are generally performed to reduce costs, material and energy, and to achieve maximum heat transfer. The main challenge in the design of the heat exchanger is to make it compact and to obtain maximum heat transfer within minimum space. Using the passive enhancement technique in coiled tube heat exchanger has considerable capacity to enhance heat transfer by creating secondary flow in the tube. The analysis of flow and heat transfer in the helical coil tube is important because it increased heat transfer. Modeling of the heat transfer properties of the shell and coil heat exchanger was described in literature. Because of high heat transfer coefficient and compact size compared with straight tubes, the helically coiled tube heat exchangers have been extensively studied as one of the passive heat transfer enhancement [26-27].

2.2 Shell and Coil Heat Exchanger

The main advantage of shell and coil heat exchangers is their simple design, low-cost production, and high efficiency. Also, shell and coiled tube heat exchanger is one of the compact heat exchangers types used to increase heat transfer rate that requires less volume and weight compared with other types of heat exchangers. Prabhanjan et al. [28] studied the comparison of heat transfer rates between a helically coiled tube and a straight tube heat exchanger at constant wall temperature and heat flux with fluid-to-fluid heat exchanger. The results showed the use of a helical coil heat exchanger increase the heat transfer coefficient compared with straight tube heat exchanger. Compared to the traditional heat exchanger, the flow field and